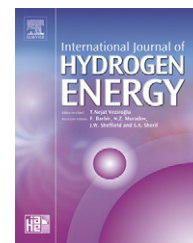


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Design and construction of a Stirling engine prototype

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ABSTRACT

In this work an external combustion engine is presented, which was designed through principles of energetic similarity and scaling, combined with adiabatic simulation, pressure loss and heat exchange characteristics analysis. These type of engines have the great advantage of having the heat source applied from outside, therefore resulting in very versatile machines which can be utilized with hydrogen, concentrated solar energy, biomass or fossil fuels. Details of the construction carried out mostly with conventional machine parts produced in Argentina are given. Finally, the results of the preliminary tests with the constructed prototype are shown.

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1. Introduction

At the present time Stirling engines are again coming to the scene because of their advantages on a fossil fuel shortage context. A Stirling motor uses an external heat source; therefore it is more versatile, being capable of adapting to different types of heat sources or fuels. There are just a few enterprises that produce these kind of engines in the world, and the supply is restricted.

The theoretical ideal cycle of the Stirling engine is shown on Fig. 1. Process 1–2 is an isothermal compression. The heat rejected on this process is absorbed at the compression exchanger by the refrigeration water. The process 2–3 is an isochoric warming in which the heat is given by the regenerator. The process 3–4 is an isothermal expansion in which the cycle gives useful work and absorbs heat from the heater. Finally, the cycle is completed by an isochoric cooling represented by process 4–1. Here, as in process 2–3, the heat is exchanged between the gas and the regenerator.

From this description it can be observed that the regenerator is a very important part of the cycle, absorbing heat in the process 4–1 and ceding ideally the same amount of heat

during process 2–3. The heat exchanged on these two transformations is up to 7 times the heat on processes 1–2 and 3–4 [1].

The design presented here is based on the energetic similarity and scaling concept [2,3]. This approach is considered to be appropriate for this project because it provides the possibility of having a fast answer with acceptable efficiency, similar to that of the prototype from which it is scaled [3]. Besides, there is few published information about the use of the scaling method to design Stirling engines, and therefore this project can provide new empirical evidence to the method. The aim of this project is the local design, construction and testing of a Stirling engine.

2. Geometrical configuration and preliminary power calculation

The power to be delivered from the engine for this project as specified on its formulation is in the range of 0.5–1 kW.

It was decided, for construction simplicity, to make an alpha-type engine converting a V-type air compressor using most of

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Nomenclature			
P	power	λ_{hxe}	normalized hydraulic ratio of expansion exchanger
N_B	Beale number	λ_{hxc}	normalized hydraulic ratio of compression exchanger
P_{ref}	reference pressure	η_v	volumetric porosity
V_{sw}	swept volume	N_{MA}	characteristic Mach number
f	frequency	N_{SG}	characteristic Stirling number
n	rpm	N_{TCR}	characteristic thermal capacity ratio
S_{th}	thermodynamic stroke	N_F	characteristic Fourier number for regenerator
λ_{xe}	normalized length of expansion exchanger	N_t	temperature ratio of hot to cold cylinder
λ_r	normalized length of regenerator	Q_e	heat exchanged on expansion space
λ_{xc}	normalized length of compression exchanger	Q_r	heat exchanged on regenerator
α_{ffxe}	normalized free flow area of expansion exchanger	Q_c	heat exchanged on compression space
α_{ffr}	normalized free flow area of regenerator		
α_{ffxc}	normalized free flow area of compression exchanger		

its mechanical parts [2]. Through this working method was expected to accelerate the mechanical design and to focus on the thermodynamic design.

For the selection of the compressor to convert, Eq. (1) (Beale equation) was used. This equation is an empirical correlation found on most of the successful Stirling engines constructed [3,4].

$$P = N_B \cdot p_{ref} \cdot V_{sw} \cdot f, \quad (1)$$

where P is the Power of the engine W, N_B is the Beale number (0.11–0.15), p_{ref} is the reference average pressure of the cycle (bar), $V_{sw} = \sqrt{2} \cdot V_{displaced}$ is the swept volume (cm^3) and f is the operation frequency (Hz).

The last three values in Eq. (1) are available in most compressor specification brochures. It is decided, also for construction convenience, that the engine will operate at the same frequency as the original machine and that the reference pressure will be equivalent to 80% of the maximum

permitted pressure on the compressor. Solving Eq. (1), with $N_B = 0.11$ for three models of commercially available compressors (2, 3 and 5.5HP all of them have two identical cylinders) gives the following values: the compressor of 2 HP gives an expected power of 420 W, the 3 HP one gives 650 W and the 5.5 HP gives 1379 W. From them, the compressor of 3 HP, which falls on the required range, is chosen.

3. Thermodynamic design and calculation procedure

As mentioned before, the engine has been designed through scaling from an engine of known characteristics and performance. The design procedure is the one described by Organ in [3, chapters 4 and 5].

As a first step, the prototype engine was chosen, from which the new design was scaled. There exist specifications of various known engines on the literature [1,3,5]. From them, the V-160 has been chosen as the model prototype, since it assures similar geometry and arrangement. On Tables 1 and 2 the parameters of the V-160 are shown.

The scaled design with the mentioned method gives the results on Tables 3 and 4.

In order to analyze the performance of this preliminary design, the “Simple” code from Urieli is used [6]. It allows to run a Schmidt analysis, an adiabatic simulation and finally a

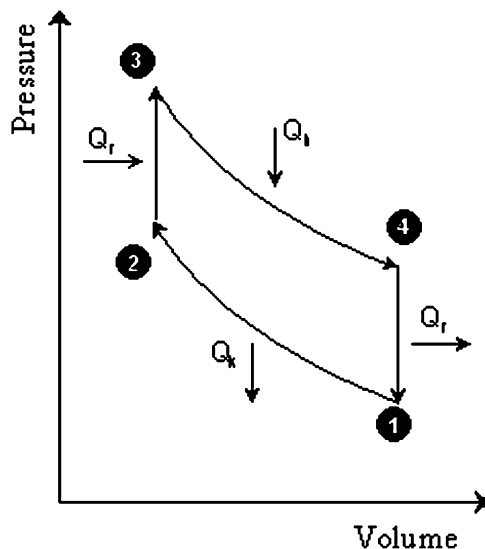


Fig. 1 – Ideal Stirling cycle scheme.

Table 1 – Operation point of the V-160 engine

Working fluid	He
p_{ref} (bar)	150
n (rpm)	1500
$P = \text{power}$ (kW)	9
$V_{sw} = \text{swept volume}$ (cm^3)	225
Thermodynamic stroke S_{th} (mm)	60.8

Table 2 – Dimensionless numbers of V-160

<i>Geometrical relationships</i>	
λ_{xe}	3.947
λ_r	0.493
λ_{xc}	1.644
α_{ffxe}	0.037
α_{ffr}	0.6188
α_{ffxc}	0.1
λ_{hxe}	0.011
λ_{hr}	0.00046
λ_{hxc}	0.00514
η_v	0.69
<i>Working conditions</i>	
N_t	2.72
N_{MA}	1.21E-02
N_{SG}	5.62E + 09
N_{TCR}	87
N_F	10.19

Table 3 – Operation point of the new design

Working fluid	He
p_{ref} (bar)	6.6
n (rpm)	620
P = power (kW)	0.65
V_{sw} = swept volume (cm ³)	902
Thermodynamic stroke S_{th} (mm)	96.6

Table 4 – Exchangers specifications for the new design

<i>Regenerator design</i>	
Length L_r (mm)	73
External diameter (mm)	83.7
Free flow area of regenerator A_{ffr} (mm ²)	3794
Wire diameter d_w (mm)	0.26
Mesh number m_w	39
Number of screens in regenerator stack	138
Thermal capacity ratio N_{TCR}	1801
Fourier number of regenerator N_F	0.89
<i>Expansion exchanger design (heater)</i>	
Number of tubes	3
Tube length (cm)	58.1
Tube diameter (mm)	9.405
<i>Compression exchanger design (cooler)</i>	
Number of tubes	41
Tube length (cm)	24
Tube diameter (mm)	4.39

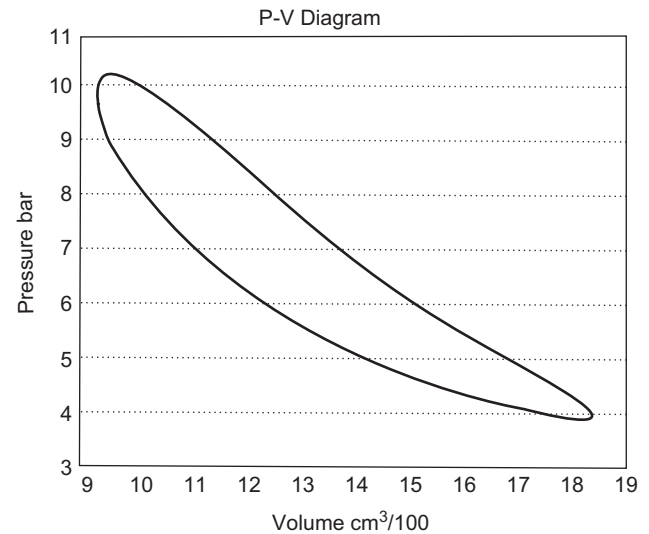
simple pressure loss and exchanger performance. As a final result, expected power and performance are obtained. The entry data for this program are some of the results obtained in Tables 3 and 4. On Tables 5 and 6 and Fig. 2 the results are exposed.

Table 5 – Results of the ideal Schmidt analysis for the proposed design

Work/cycle (J)	152
Q_e Rejected heat per cycle by exp. exch. (J)	241
Q_c Absorbed heat on comp. exch. per cycle (J)	88
Power (W)	1520
Indicated efficiency	0.632

Table 6 – Analysis results

<i>Adiabatic analysis results</i>	
Q_e (W)	2818
Q_c (W)	1400
Power (W)	1423
Thermal efficiency	0.505
<i>Results of the exchangers analysis</i>	
Reynolds number for heater	6659
Reynolds number for cooler	2447
Reynolds number for regenerator	31
Number of transference units on regenerator	40.8
Regenerator efficiency	0.976
<i>Theoretical expected efficiency and power loss</i>	
Pressure loss (W)	56.5
Power output (W)	1097
Power input (W)	2962
Efficiency	0.37

**Fig. 2 – P–V diagram from adiabatic analysis.**

4. Construction

Tables 3 and 4 contain the specifications with which the construction of the thermal circuit was performed. From the original compressor almost all its parts were used: crankcase, flywheel, crankshaft, con rods, pistons, piston rings and cylinders. Few parts were modified.

The compression exchanger tubes are made of copper and were soldered to the tube holder plate. The regenerator was

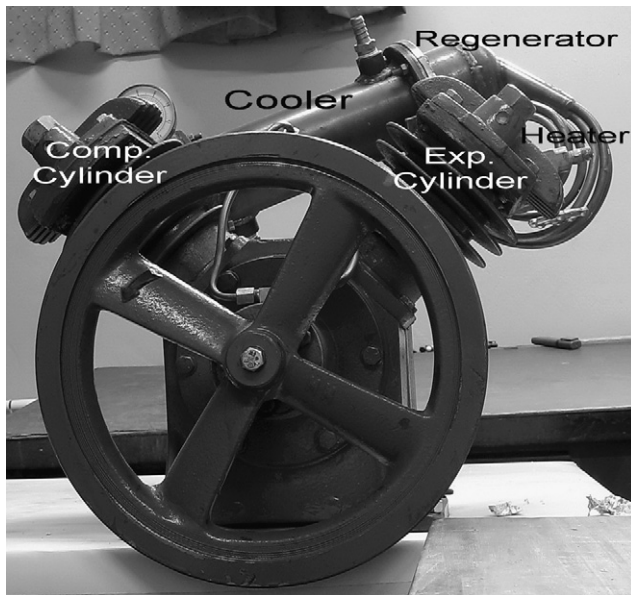


Fig. 3 – Complete engine with detail of its components.

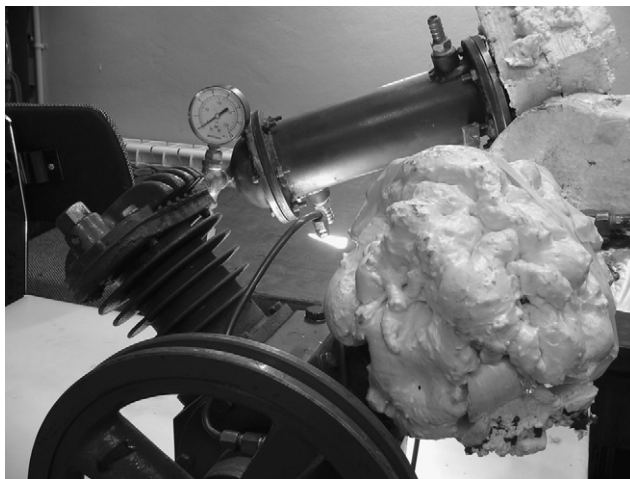


Fig. 4 – Test bench with the equipment mounted. On the upper left corner the regenerator and the expansion exchanger covered by thermal insulation can be seen.

constructed from stainless steel mesh number 35, being the one closer to the specifications found in local market. The 138 circular mesh plates were cut and were located adjacent to the compression exchanger between cooler and heater on Fig. 3.

On Fig. 4 the complete device is exposed. The thermal circuit is at the rear part on the picture. Next to each cylinder charge and discharge valves were placed. The original valves and air inlets of the air compressor were eliminated.

5. Preparation and results of the first experimental test

With the aim of measuring the main working parameters of the equipment, a previous test as a refrigerating machine was proposed as it was done before by other authors [7]. As it is

well known, the Stirling cycle is very simple to reverse. This way, heat is extracted at the expansion exchanger at a low temperature and ceded at the compression exchanger at a higher temperature. The necessary work is supplied to the cycle by an electrical motor coupled to the flywheel, Fig. 4.

The advantage of the method used to carry out the test is basically that it allows to obtain performance data even when the machine has too low performance to achieve functioning itself as a motor. Furthermore, that permits to obtain numerical results when executing redesign and improvements on the machine.

The tests were performed at a rotational speed of 200 rpm with a 1 HP electrical engine and the initial pressure of helium in the circuit was 7 bar.

It is important to point out that these tests were performed at lower speed than that from design, and represented just one point of operation.

The measured parameters during the experience were the outer temperature of expansion space T_e , in/outlet water temperature to/from the compression exchanger and equilibrium temperatures on inlet and outlet of regenerator case. The measurements were done with type K thermocouples.

6. Results

On Figs. 5 and 6 some experiences results are graphed. In Fig. 5 cooling curves of expansion exchanger can be seen, while on Fig. 6 the regenerator temperature behavior on cold side can be observed.

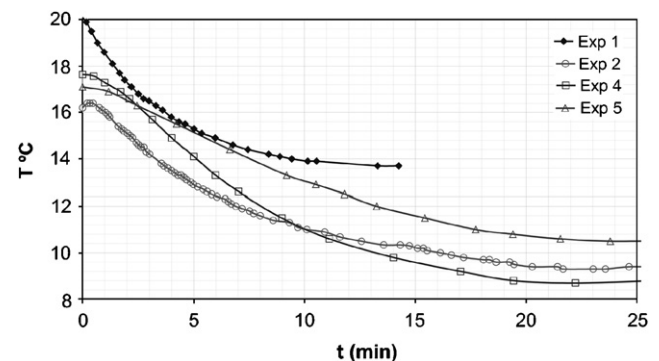


Fig. 5 – Cooling curves of different experiences.

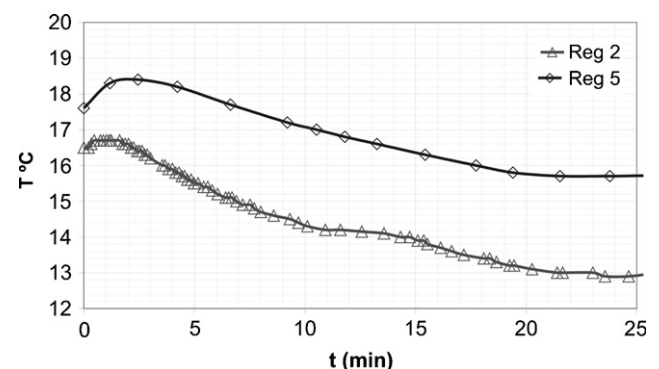


Fig. 6 – Regenerator temperature curves for two experiences.

The experience 1 of Fig. 5 was carried out without circulation of the refrigeration water in the compression exchanger. The other ones were obtained with a water flow of 1.3 l/min. The experience 5 shows a lower rate of temperature decreasing. The reason for this is that it was done just after the previous experience and the initial temperature of the complete equipment was different.

The cooling curves of the regenerator cold side next to the expansion space show a coherent behavior with the expansion exchanger, although with more inertia and less temperature drop, Fig. 6.

7. Conclusions

First, the preliminary results verify that the design meet the needs of its working principle, extracting heat from the lower temperature source and giving it to the higher temperature source. Besides it brings useful information of the proposed design, whose performance is still very low and must be improved.

On the cooling curves it is observed that after a transient state, temperature begins to diminish and it stabilizes itself in approximately 25 min.

On the other hand, from the measurements of the inlet and outlet water temperature it is determined that the heat given to it corresponds to a power of 140 W in average.

8. Future perspectives

It must be mentioned that this project is part of a research program of the National University of Cuyo. In this project the

Stirling engine is part of a dish-Stirling system which would be installed in the north of Mendoza.

The results obtained up to now are encouraging to continue. It is foreseen to improve and redesign each part of the engine and eliminate dead spaces of the exchangers.

Acknowledgments

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